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# THE SLIDE VALVE

BY

N. P. BURGH.

# THE

# SLIDE VALVE

## PRACTICALLY CONSIDERED.

BY

N. P. BURGH, ENGINEER,

AUTHOR OF "A TREATISE ON SUGAR MACHINERY;"
"PRACTICAL ILLUSTRATIONS OF LAND AND MARINE ENGINES;"
"A POCKET BOOK OF PRACTICAL RULES FOR DESIGNING
LAND AND MARINE ENGINES, BOILERS, ETC."
ETC. ETC. ETC.

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1865.

186. f. 8.

# THE SLIDE VALVE

BY

N. P. BURGH.

# PREFACE.

THE relative position of the engineer to the steam engine may be said, with some truth of comparison, to be analogous to that of the medical man and his patient. Of all the details (so to speak) of the human body, the heart is the most important. The same may be said of the slide valve, in comparison with the remaining details of the engine. In either case the defective action of the heart or valve impedes the progress of the whole.

The present work is intended by the author to enable the student and interested observer to clearly understand the use and action of the slide valve. The rules relating to the equilibrium slide valves have been carefully digested; that portion relating to the "width of the large bar" is, perhaps, the most important of all, while the simplicity is perfect. The delineation of the eccentric circle should he well studied, it being remembered that from it the slide is proportioned. The angles assumed by the cranks at given grades of expansion are matters too often disregarded by the pupil and young engineer. It is not usually appreciated that the arc passed through determines the lap of the valve. In no case whatever

should a valve be designed without first considering the direction of the crank pin, and the amount of expansion required. The proportion of lap to width of opening or port can only be obtained by the means alluded to. The lead of the slide valve has been duly considered in this work. The author would impress that the true versed sine and radius of the arc of supply steam can only be attained, *by noticing the lead.* The time for supply, expansion, exhaustion, and compression has been fully explained, so that its value and defects will be apparent. The delineation of the crank circle has met with due attention, and will be found of utility in proving the actual consumption of the steam at each stroke of the piston.

In the chapter relating to "General Observations" will be found hints worthy of notice. The proportions alluded to are deduced from the best results.

The index will be found lucid and copious, so that the allusion made to any detail in requisition is rendered perceivable *per se.*

The author has carefully avoided theorizing on the subject on which he treats; all his remarks have been deduced from *practical* demonstrations —assumptions have *not* been admitted—therefore the data given will be found exact.

N. P. BURGH.

78, *Southampton Terrace,*
*Waterloo Bridge,*
*London, October,* 1865.

THE

# SLIDE VALVE PRACTICALLY CONSIDERED.

## CHAP. I.

### ANTECEDENTS OF THE SLIDE VALVE AND STEAM PORTS IN THE CYLINDER.

THE steam engine of the past and that of the present widely differ in the consumption of the fuel, or rather steam, per indicated horse power. Twenty years ago, eight to ten pounds of coal per actual horse power was deemed a good result, while in the present day two and a half pounds of coal per horse power is considered but a fair attainment. This great reduction is due to two causes,—perfection of manufacture and arrangement, and the application of natural laws.

The friction due to the surfaces in contact of all bodies are alike in principle. The slide valve may be said to be a body moving on a given area, the pressure of the steam being the load on the same. An idea of the friction imposed can be attained by

multiplying the pressure of the steam per square inch by the area of the surface operated on. This result, multiplied by the co-efficient of motion (the unguent being of course considered), will give the power required to move the valve when under full steam. Many ideas have been promulgated and carried into effect to neutralize the action of the steam on the valve. Some of the means adopted are,—the valve is hollow centrally, or as a frame; the surfaces at the back and face are equal, hence the friction is due only to those parts in contact. Adjustment is attained by a wedge-like division, a band and set studs being introduced to prevent lateral and longitudinal looseness. In another example, the action of the steam preserves the required amount of adjustment, so that the details in the former example are dispensed with.

The lead of the slide valve greatly affects the lap of the valve, this can be understood by remembering that, when the crank is on the horizontal line, the valve is open, due to the lead required. The valve travels back to the extremity of its stroke, and forth to close the port, while the crank pin is traversing through a given arc. The grade of expansion attained, is due to the radius of the arc, passed through by the eccentric centre.

This is fully investigated and explained in the fourth chapter. The rule given for the outside lap, in page 43, does not consider the lead, should the latter be required to be introduced, the true versed sine of the eccentric arc of supply steam, would equal *width of the opening caused by the valve, minus the lead.* In some instances half the lead must only be deducted, while in other examples, the first rule alluded to is correct. This diversity is due to the length of the eccentric rods, and locality of the valve and gear, &c., allusion will now be made to each portion *seriatim.*

The mode of setting out the steam ports has often been a mystery to the many and thus known only to the few. There is, however, no just cause for this inequality of information; there is no reason why the fitter, who lines out the ports on the cylinder facing, or that for the valve, should not clearly understand the cause and effect. Let it be supposed an engine is to be constructed, and the lap of the valve given, to cut off a given portion of the stroke; in most cases the fitter engaged thereon, or even, perhaps, the foreman, does not understand the reason, or rather natural cause, for the dimensions to be adhered to. Now, were this matter more often considered and the practical observer initiated into the simplicity (not

mystery) of the action of the slide valve, doubtless a general improvement would be the result.

Presuming that there are a few who may be ignorant of the true proportion of the ports in the cylinder, and the action of the slide valve on the same, therefore, a concise explanation will not be out of place. The ports in the cylinder are for two purposes—supply and exhausting the steam. Now it is obvious that the pressure of the steam, on entering the cylinder, is greater than when exhausting therefrom. The proportion of the supply port to that for the exhaust is not a fixed sum or rule; as in all mechanical operations, circumstances alter cases—suffice it to say that in the case of the ports the same must be observed.

The area of the cylinder is the first consideration that engrosses the attention of the designer. Now it may be said that there should be a fixed rule for this, but practice teaches that the pressure of the steam should be considered. Many high-pressure engines greatly differ in the diameter of the cylinders, area of the steam ports, &c. The cause for this variation is the different ideas on the subject. In some instances the 12-horse power in one county is only ten in the next, and *vice versa.*

Presuming that a fixed pressure of steam is observed, and speed of piston agreed on, the areas of the cylinder and ports are matters of simple calculation. Some engineers prefer the areas of the ports to be decided by a complex rule rather than allow the area of the port to be deduced from a given number of square inches per nominal horse power. It must be understood that when expansion is required the cubic contents of the cylinder should be increased; in order to attain this the area is enlarged. Now, were the port deduced from the said area, the same would be increased in proportion. Steam, under a given pressure and capacity, occupies a certain space; should the latter be increased the former is decreased in proportion. It will be thus seen that the better formula for the area of the ports should be number of square inches agreed on $\times$ horse power nominal. Should a fixed rule be determined on for that of the cylinder, including a given pressure and speed of piston, a similar result can be attained by, $\frac{\text{area of cylinder.}}{\text{constant number.}}$

The author's extended experience in engines of all classes, embracing all the modern improvements by the best makers, enables him to give the following correct formula for the area of the

opening or port caused by the valve for the supply steam :—

High-pressure engines = H.P. × ·6 to ·5 sq. in.
Condensing engines = „ × 1. to ·75 „

To arrange the ports, after having determined the area, is first to determine the width and length. The shape of the ports are always that of a parallelogram, hence the area divided by the length equals the width, and *vice versa*. It may not be out of place to state that the length of the port should be in proportion to the diameter of the cylinder. In setting out the ports, three important facts must be considered:—ratio of supply to exhaust, width of bar, and outside lap of the valve. The first is determined from the speed of the piston and pressure of the steam; the second from the thickness of the metal of the cylinder; and the last is due to the stroke of the engine, length of connecting rod, position of the valve in relation to the connecting rod, and the grade of expansion required; indeed, it may be truthfully said that the outside lap of the valve is subservient also to design and arrangement—this will be better understood from future allusions and illustrations. The proportions herein given are taken from practice, and therefore may be relied on.

The present modes adopted for reducing the pressure on the back of the valve, are simply a ring encompassing a raised portion on the valve; a fitting projection on the inside of the case enables a perfect joint to be made; and a communication with the condenser causes a vacuum at the back of the valve, in place of the pressure of the steam. For small valves, springs are used to assist the pressure, the area being small there is no cause to reduce the pressure at the back.

The means for securing the valve rod to the valve are numerous; some makers prefer a clip or ring, others adopt a screwed fixture; a third will put the rod through the valve, while a fourth authority will make a solid T head to the rod, and fit the same in the valve. The best mode is to secure the rod into a loose nut of gun-metal, sufficient play being allowed for wear and repairs. In some instances two rods are adopted.

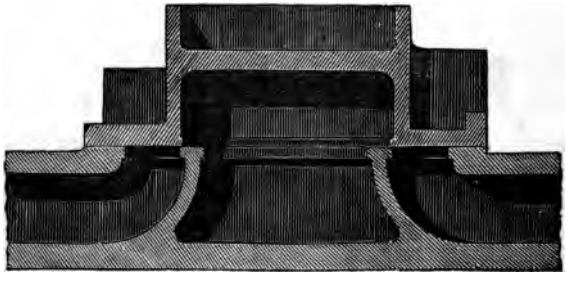
## CHAP. II.

### THE PROPORTION OF VALVES AND PORTS IN THE CYLINDER. COMMON AND EXHAUST RELIEF SLIDE VALVES.

SLIDE valves of the present day are divided into three classes or kinds—common, exhaust relief, and equilibrium. It is now proposed to describe the two former.

The common slide, Fig. 1, will be readily understood by the diagram, as to its shape, &c. As,

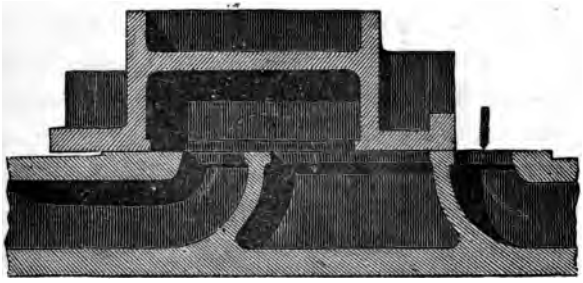
FIG. 1.



perhaps, it will not be out of place to give examples for future reference, allusion will be made to the proportions of the valve relatively

to those of the ports of the cylinder. Let it be presumed that a valve, &c., is required for an engine of 65-horse power nominal, adapted for the screw propeller. Now, allowing 17·5 square inches per horse power, the cylinder would equal about 38in. in diameter, having a stroke of 2ft. The size of the steam port supply being generally 1 square inch (in area) per horse power, the result would be 65 square inches. Thus far having determined this proportion, it now remains to decide the proportion of the ports, &c., not at a guess, but by correct data, certain for all dimensions or ratios. It will be noticed in the diagram, Fig. 2, that the slide, when at full

FIG. 2.



stroke, moves the full width of the port, hence the term common slide. Now, it is obvious that the area of the port being given, that same being

divided by its length, will produce the requisite width, which latter, in the present example, will be 2½in., outside lap (presumed) 1½in., inside lap ⅛in.; having thus far agreed, the following rules will supply what is required:—

Width of bar = width of port × ·5, but 1in. as a minimum.

Width of exhaust space in valve = width of port supply × 1·5 + half travel of the valve + width of bar, minus inside lap.

Width of exhaust port in cylinder = width of bars minus inside laps, deducted from the width of exhaust space in the valve.

Now, as the two last rules may seem a little confused, it will be well to explain their origin. The exhaust space in the valve is that part that allows the steam to escape into the waste port. It is the general practice that the area of the exhaust port should exceed that of the supply; some engineers prefer supply 1, exhaust 2; others, the former 1, the latter 1·5. It is clear that when the valve is at full stroke, as in Fig. 2, one bar only is in question as to the area for the exhaust; the half travel of any valve, it is almost needless to say, is the outside lap, plus the width of the supply port; thus it will be understood that the inside lap must be also taken into

consideration to produce the exact proportion, although it may be said this last is not of vital importance, yet to attain the correct result, it must be noticed.

For explanation of the last rule, Fig. 1 will readily convey an idea of its origin and the truth of its composition; the inside laps, it will be seen, determine the dimension required. Simple as this rule may appear, it is of as much importance as those preceding it, more especially when requiring the width of the exhaust port by calculation in the absence of a diagram.

It is not here presumed that there is any new formula given; on the contrary, the author rather wonders that such a simple and correct mode of producing the proper proportions or distances is not more often alluded to. Many young engineers of clear perception look on this portion of the data requisite for the steam engine as the most difficult of all; but first determining the ratio of the length of the connecting rod to the stroke of the engine, and the grade to cut off the steam at, the remainder may be said to be a simple fact. Yet, withal, there is, perhaps, an atmosphere of mystery pervading the region alluded to, only to be dispelled by practice and thought. It may as well be added that

the two proportions last mentioned will be reverted to *seriatim*. Returning to the requisite dimensions for the example given, they will be found to be thus:—

Let A = width of port supply.
B = outside lap.
C = inside lap.
D = width of bar.
E = width of exhaust space in valve.
F = width of exhaust port in cylinder.
G = half travel of the valve.

| | | Inches. |
|---|---|---|
| Then, | $A = \frac{65}{26} =$ . . . . . | 2·5 |
| | $B = 2{\cdot}5 \times {\cdot}6 =$ . . . . . | 1·5 |
| | $C =$ . . . . . | ·125 |
| | $D =$ . . . . . | 1·0 |
| | $E = 2{\cdot}5 \times 1{\cdot}5 + 1{\cdot}5 + 2{\cdot}5 + [1 - {\cdot}125] =$ | 8·625 |
| | $F = 8{\cdot}625 - [1 - {\cdot}125 \times 2] =$ . . | 6·875 |
| | $G = 2{\cdot}5 + 1{\cdot}5 =$ . . . . . | 4·0 |

The above calculations, and the results, represent the proportions of the valve and ports in Figs. 1 and 2. It may, perhaps, be well to add, that the formula given will be correct for any proportion, whether the valve is intended to cut off at one-quarter, three-eighths, one-half, five-eights, three-fourths, or seven-eighths of the

stroke. Also the exhaust may bear any proportion to the supply, remembering, of course, to use the ratio required, rather than the one given at present.

In Fig. 2, the valve, as before stated, is at full stroke. It will be seen also that the end of the valve overlaps or travels beyond the supply port (exhausting). This overlap will be found, in all cases, to equal the outside lap, minus that of the inside. Therefore, when the valve has closed the supply port, the exhaust will be open in width, equal to the outside lap minus inside lap.

It needs scarcely be added, that when no inside lap is given, the outside lap regulates the excess of time (beyond that for the supply) for exhausting. In the example now alluded to, the areas (from the supply ports) both for exhaust and supply, are equal; due, of course, to the slide travelling the entire width of the port. Now, to lessen the stroke of the valve would be to reduce the area for the *entrée* of the steam, while that for the exhaust would not be affected, *i.e.*, should the reduction of the half-stroke of the valve not exceed the overlap on the exhaust side in Fig. 2. To retain the area for the *entrée* of steam—but increase that for the exhaust more in proportion than in the diagrams Figs. 1 and 2, but not to

exceed or reduce the stroke of the valve—the width of all the ports must be increased in proportion, hence the origin of the "exhaust relief valve."

For the purpose of future comparison and illustration, it will be better to give the proportions and formula for this valve. Now, presuming an engine of the same power as the last example, and ratio of area for the supply steam to be equal, also the stroke of the valve—but allowing the area of the port to exceed that of the opening (caused by the stroke of valve)—the following result is obtained, viz.:—The exhaust side of the piston is less susceptible to the action of the steam, generally known as back pressure or "cushioning;" the explanation of this will be further alluded to hereafter. The following formula or data will now be in requisition before proceeding further:—

Width of port = width of supply opening caused by valve × 1·5.

Width of exhaust opening caused by the valve = width of port × 1·5.

Width of exhaust space in valve = width of port × 1·5 + half travel of the valve + width of bar minus inside lap.

Width of exhaust port in cylinder = width of

bars minus inside laps deducted from the exhaust space in valve.

Before making use of these rules, it will be

FIG. 3.

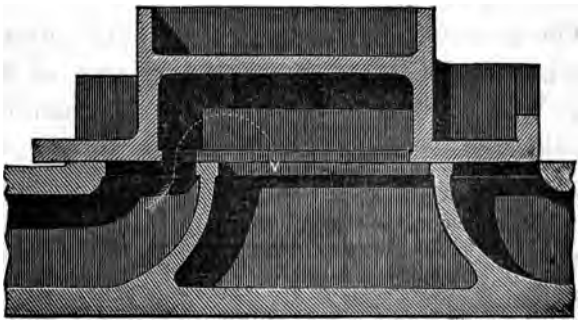
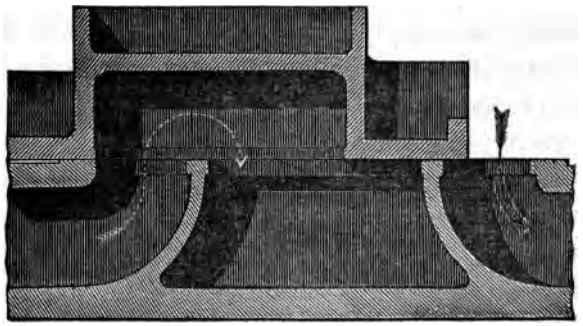


FIG. 4.



requisite (to prove their veracity,) to explain their origin. To begin with the rule for the width of the port, it will be seen that a constant

number is given. Now, it must be strictly understood that this sum can be lessened or increased according to the discretion of the calculator, the number given being compiled from the average of general practice.

The second rule bears strict reference to the distance of the inside of the bar to that of the valve when at full stroke, as at Fig. 2, hence the increase of area or width of the exhaust port to that of the supply. The width of the exhaust space in valve is produced by a formula, as in the first example; also the last rule now given is the same in composition. In Fig. 3 will be seen a diagram of a valve and ports arranged for the exhaust relief principle. The valve is entirely covering one supply port, or cutting off, while the exhaust side is partially open; due, of course, to the arrangement before alluded to. In Fig. 4 the valve is shown at full stroke. Now, on comparing the widths of the ports (for the exhaust) in Figs. 2 and 4, the relative areas can be readily understood (the scale being the same throughout). It is, therefore, obvious that, while the valve is opening the port, for a given width as in Fig. 4, that for the exhaust is increased, the ratio being due to the discretion of the designer. Having thus briefly described the action of the valves

Figs. 3 and 4, for the better purpose of practically proving these statements, attention will now be given to the calculations. As before stated, this valve is for engines of the same nominal horse-power as in the previous example.

Then, let—

A = width of opening caused by the slide.
B = width of port supply.
C = outside lap.
D = inside lap.
E = width of bar.
F = width of exhaust space in valve.
G = width of exhaust port in cylinder.
H = half travel of the valve.

Let $A = \frac{65}{26} =$ . . . . . 2·5

$B = 2{\cdot}5 \times 1{\cdot}5 =$ . . . . . 3·75

$C = 2{\cdot}5 \times {\cdot}6 =$ . . . . . 1·5

$D =$ . . . . . ·125

$E =$ . . . . . 1·0

$F = 3{\cdot}75 \times 1{\cdot}5 + 2{\cdot}5 + [1{\cdot}0 - {\cdot}125] = 10{\cdot}5$

$G = 10{\cdot}5 - [1{\cdot}0 - {\cdot}125 \times 2] =$ . . 8·75

$H = 2{\cdot}5 + 1{\cdot}5 =$ . . . . . 4·0

On referring to Fig. 4, the relative value of the above dimensions can be better understood. The supply opening is opened 2½in., but the port on

the opposite side is opened full or $3\frac{3}{4}$in., and the distance from the inside of the bar to that of the valve will be $5\frac{5}{8}$in., or the width for finally exhausting; thus it will be understood the ratios of the ports to each other, is as 1 to 1·5. Now this ratio is not imperative, 1 to 2, or 1 to $2\frac{1}{2}$, or 1 to 3, may be adopted, or as in Figs. 1 and 2, the ratios may be equal as far as regards the ports and openings.

It will now be well to allude to the lead on the exhaust side. It should be understood that the only means of creating a perfect indicator diagram (the best mode yet known to indicate the action of the steam) is to cut off suddenly and exhaust freely. Should the latter not be accomplished the back pressure or extreme cushioning, before alluded to, will impede the action of the engine. It is, of course, well known that a lead on the steam side is often given, which will, to a certain extent, produce a cushion, or a round corner, so to speak, in the indicator diagram; but when this is intentional, the evil may be said to be admissable. It will be seen, in Fig. 3, the lead of the exhaust is equal to the outside lap of the valve minus the inside lap. The same applies to Figs. 1 and 2, although not represented. It is, therefore, plain that the lead on exhaust side (in

the present rules) is due to the outside lap minus that of the inside.

The adoption of the narrow bars is correct both in practice and theory, the friction being reduced in proportion to wide bars. The larger area also is maintained for exhausting the steam freely for a given length of valve, or stroke of the same. In Fig 1 the valve is represented at half stroke. Now, it can be readily understood that the valve has to move the remainder of the width of the bar (on the exhaust side) before the area of the exhaust port is contracted; therefore, the steam can fairly exhaust at its greatest pressure through the larger area, both pressure and area being reduced proportionately, so to speak.

With reference to the inside lap of the slide valve, its vital intention is to ensure that the ports are covered on the exhaust side, so that no leakage can ensue when the valve is at half stroke.

## CHAP. III.

### EQUILIBRIUM AND DOUBLE PORTED VALVES.

In the days that the side-lever engine rejoiced in being supreme, the slide valve was deemed necessary to have a stroke from 6in. to 24in., 18in. being the average for engines of considerable power. Now, it is obvious that, when two bodies are in contact, the longer the sliding motion imparted is continued, the greater the friction; due, of course, to the surface in contact and pressure imposed; but should the surface and the motion be reduced, the friction is likewise affected. Double beat valves have been adopted for slow-speed engines, but are not, as at present arranged, applicable for the marine engine of the present day. The common short slide valve, as previously described, has been long in use for engines of all powers and classes, but the friction, &c., has greatly tended to retard its general adoption —hence the introduction of the double ported valve, commonly known as the "equilibrium or gridiron" slide valve. The term equilibrium is

due to the peculiar shape of the body part of the valve, the steam thereby being allowed to act on the external and internal portions at the same time, equally or unequally, as the form will admit.

This arrangement is not entirely novel. It will be remembered, doubtless, by many, that the slide valves used for the old table and other engines were hollow between the faces; the supply steam acting on the exterior portion, and the exhaust on the interior, thus causing an equalisation of the action of the steam, due to that portion it surrounded. The equilibrium slide valves of the present day are not much varied in principle, the difference being due, rather to proportion than design.

Fig. 1 is a good example, properly proportioned, both for supply and exhaust relief. It will be readily understood that the stroke of this valve will be exactly half of that of the common slide; due, of course, to the double ports in the place of single. Another great advantage in the reduction of the stroke of the valve for the same power is, in the proportions of the eccentric rods and links in a given space, particularly where direct action is imperative. The frictional surface is, perhaps, increased in the present example

FIG. 1.

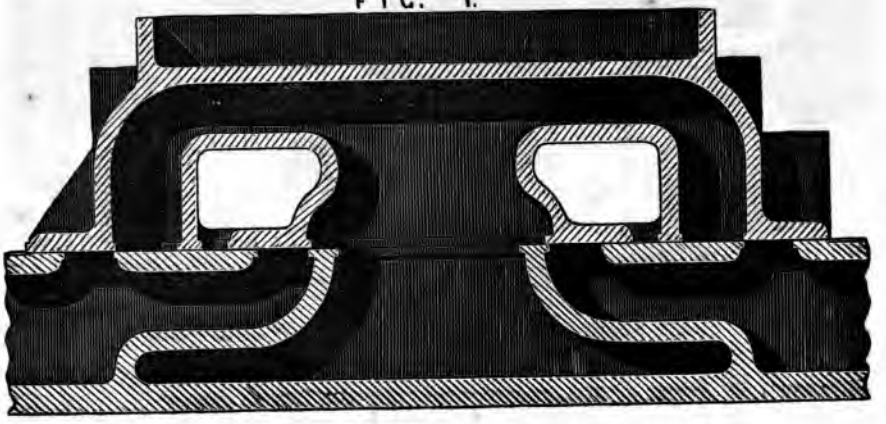
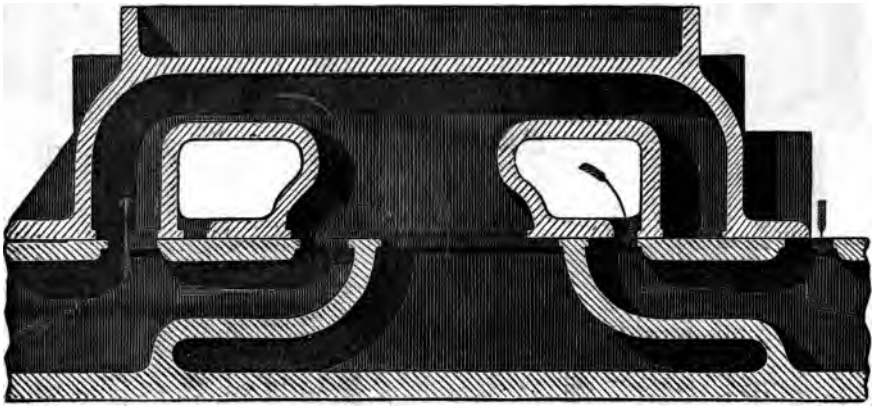


FIG. 2.



as compared with that of the common kind; but this evil is counteracted by the reduction of the stroke, and partial neutrality of the action of the steam on the valve. The valve (seen in Fig. 1) is at half stroke or covering all the ports. Fig. 2 will be understood to illustrate this valve at full stroke; the supply steam is indicated by full line arrows, while that for the exhaust is represented by the dotted kind, as in the preceding diagrams.

The practical mode to ascertain the dimensions for a given valve has often been by diagrams, cutting and carving as the case may require. Now, presuming the absence of diagrams and the facility for producing any, calculations *properly founded*, would produce a correct result, without fear of doubt as to the practical value of the dimensions attained. In order to prove the truth of this statement, presume that the diagrams on pages 26 and 27 illustrate a valve and ports for a cylinder of 50-horse power nominal. Let it be assumed that the width of the opening caused by the slide is 1in., the outside lap to be (say) $1\frac{1}{2}$in., the inside lap to be $\frac{1}{16}$in., and the width of the narrow, or inside bar, to be $1\frac{1}{4}$in. Having decided on these arbitrary dimensions, it next becomes needful to deduce the remaining proportions.

Now, for engines moving at high velocities, it is necessary that the exhaust should have a lead on the supply, which will, of course, as before stated, be due to the width of the outside lap, minus inside lap. The proportion of the ports supply to that for the openings is not imperative. The same may be said of the exhaust port in relation to that for the supply. Presuming the width of the latter to be 2in. each, the following formula will supply the desiderata.

Width of exhaust space in valve = width of two ports supply × 1·5 + half travel of the valve + width of small bar, minus inside lap.

Width of exhaust port in cylinder = width of small bars, minus inside laps deducted from the width of exhaust space in the valve.

Width of large bar in cylinder = outside lap + width of opening caused by valve + width of small bar + half travel of valve.

To test the truth of these rules will be to disarrange all the proportions, but before entering into that subject it will be well to explain the reason for the present constant numbers. The width of the exhaust space in the valve (on referring to Fig. 1) will be understood to be based on two fixed proportions, viz., ratio of the supply

ports to that for the exhaust, and the half travel of the valve. On referring to Fig. 2 it will be seen that one bar only affects the exhaust from the outside port, hence the allusion in the rule to one bar. The last portion of this rule may seem superfluous, but when a certain dimension, or rather proportion, is fixed upon, the inside lap must be considered. Another potent reason for its introduction is, that the valve, when at full stroke, partially covers the centre or exhaust port in the cylinder (see Fig. 2). It is, therefore, imperative to attain a correct or fixed ratio of supply to exhaust, to observe the whole of the formula given.

The next rule, "width of exhaust port in the cylinder," is based on the result of the first formula. On referring to Fig. 1 this will be apparent, and the relationship readily understood. It will be better to add that narrow bars are not imperative, although now generally introduced; and that were wide bars used, the proportions last alluded to would be the same.

The third rule now under notice is the width of the large bar. This dimension is the most important of all, as from or on it must be set out the openings, or ports and bars, in the valve. On referring to Fig. 1 this may seem a simple pro-

cess; but in the absence of a diagram a correct formula is invaluable.

To return to Fig. 1, it will perhaps assist the mind more readily in understanding the principles on which this formula is based. Now, let it be presumed that the valve is removed from the face of the cylinder, the plain surface will be, perhaps, perplexing, and the reasons for the fixed dimensions deemed dogmatical. On glancing at Fig. 2 the truth will be explained. For example, the large bar in the present illustrations is $6\frac{1}{4}$in. (deduced by the formula). In Fig. 1 the valve is at half stroke. Let it be presumed that the act of testing the truth of the dimension is in operation. Commencing on the inside part of the bar, set off on it towards the outer side, or from the centre line, the outside lap. From this last point set out the width of opening caused by the valve; next the width of the small bar—this last according to discretion. From this point set off the half travel of the valve, and the result will be the dimension alluded to. Thus far, having proved the basis of the formula for a given dimension and proportions, it may with justice be argued that, as only one example has been introduced, a doubt may still be allowable. To further demonstrate by different proportions

would at present be confusing, until those alluded to are put in *seriatim.*

Then, let—

A = width of opening caused by the valve.
B = outside lap of valve.
C = inside lap of valve.
D = width of port supply.
E = narrow bar.
F = half travel of the valve.
G = width of exhaust space in the valve.
H = width of exhaust port in the cylinder.
I = width of large bar in the cylinder.

| | Inches. |
|---|---|
| Then A = (presumed) . . . . . | 1 |
| B = . . . . . | 1·5 |
| C = . . . . . | ·0625 |
| D = . . . . . | 2·0 |
| E = . . . . . | 1·25 |
| F = 1+1·5 = . . . . . | 2·5 |
| G = 2+2×1·5+2·5+[1·25−·0625] = | 9·6875 |
| H = 9·6875—[1.25—·0625×2] = | 7·3125 |
| I = 1·5+1+1·25+2·5 = | 6·25 |

These calculations and their results bear strict reference to Figs. 1 and 2. It will be seen that (in Fig. 2) the exhaust passage in the valve is wider than the port supply, and also overlaps the same. Now, this ratio of passage to ports is par-

tially due to that of the half travel to the port. It will be well to add that the correct overlap (if any) can be known by F—[D+C].

As before stated, the proportion of the large bar is the most important. Further remarks may further assist elucidation. The overlap on the large bar will, of course, be equal to that already alluded to. The constant number given in the formula for the large bar is not a fixed sum. Any proportion of lap to the width of the port or opening, to cut off at a given stroke of the piston, can be maintained. It is almost needless to add that the overlap will be increased in proportion also. For example, let it be presumed the value of

| | Inches. |
|---|---|
| A = . . . . . | 1 |
| B = . . . . . | 3 |
| C = . . . . . | 0·125 |
| D = . . . . . | 2·0 |
| F = . . . . . | 4·0 |

Then the overlap = F—[D+C] = 1·875, increasing the large bar in correct proportion also.

Now, as the two examples given are in proportion to each other, *i.e.*, with an excess of overlap, and lead for the exhaust; also the large bar exceeds the total of A, B, C, D, and E; due, of course, to the proportions decided on.

Presume a valve to be proportioned so that there shall be no overlap, (but this is given for comparison only, not as a correct proportion). In order to make this more concise as well as practicable in all cases, the following formula will greatly assist. Let F = D+C, commencing then with this datum, let

| | Inches. |
|---|---|
| A = . . . . . | 1 |
| B = . . . . . | 1·0625 |
| C = . . . . . | 0·0625 |
| D = . . . . . | 2·0 |
| E = . . . . . | 1·25 |

Then I = A+B+C+D+E = 5·375.

Again, I = B+A+E+F = 5·375.

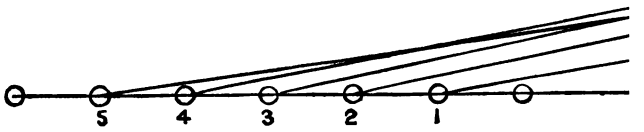
By this it will be seen that the results are alike, thus proving the truth of the concise formula alluded to. It is well known that, in some cases, for practical reasons, the laps and leads of the valves now under notice are unequal, in order to preserve a correct action. To attain this the width of the steam supply and exhaust ports in the valves, as well as the large bars in the cylinder, would be varied, but the formula here given would not be materially affected.

## CHAP. IV.

### THE PROPER MODE OF ATTAINING THE CORRECT AMOUNT OF LAP.

It is well known that the action of the steam (in the cylinder of an engine,) when cut off, is merely expansive; also that the power exerted is in proportion to the pressure of the volume admitted, and the space occupied by the same. The indicator diagram imparts a fair, and, as far as mechanical contrivances will admit of, correct illustration of the power attained. The strictly correct shape of the expansion curve is patent to all, but the causes of the diversion from the true line are often a matter of controversy. Now, in the present day our marine engines work at high velocities, consequently the time allowed for the exhaustion of the steam should greatly exceed that of the supply. The outside lap of the slide valve, as at present arranged, regulates the cut off of the steam at a given part of the stroke of the piston. A slide without sufficient lap is conducive to a waste of steam; hence the too often reduction of power in proportion to the fuel ex-

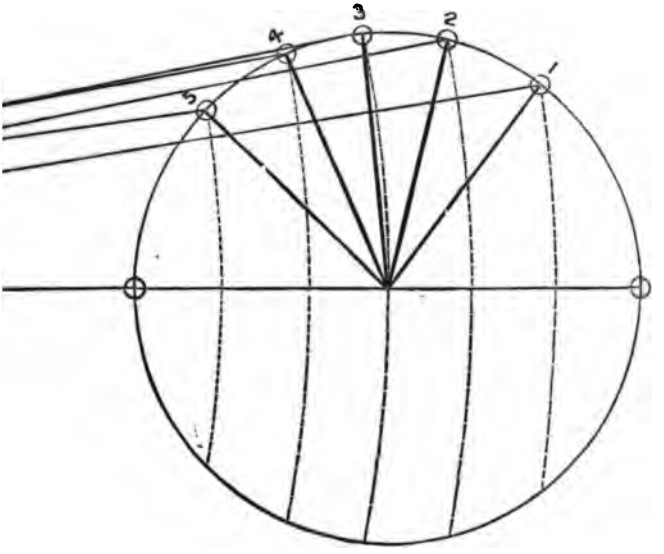
pended. Truthful as this is, yet the importance of the outside lap is too often disregarded, or treated in some instances as a matter of minor consideration, (let it be understood this remark is in allusion to the student rather than the adult members of the profession.) The author, some years ago, had the opportunity of repairing an engine, the slide valve of which had no lap (so to speak,) a fact which rendered clear the cause of the incapacity of the engine. On adding to




the length of the valve the result was quite satisfactory. Not long since a similar result was attained by reducing or plugging the width of the ports' supply, the stroke of the valve being unalterable. Our Cornish engineers pride themselves—and with justice, too—on the economical working of their engines. The valves usually adopted are the double-beat type; but the secret of—or, rather, cause of—the gain in economy is in the shape of the cam that lifts or causes the motion of the valve. Now, in the case of the eccentric and the common slide, the motion of

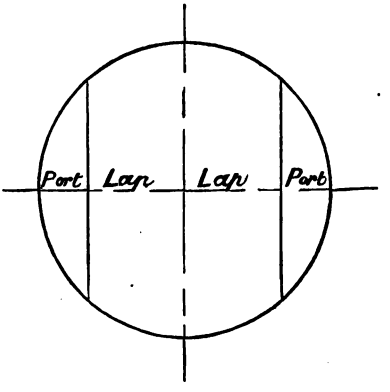
the valve may be termed uniform, therefore the outside lap *must* be considered.

The rules given by many authors for defining by calculation the amount of lap due to the grade and stroke of the engine are useful, and,




no doubt, are of value to a certain extent. Rules, however, if complicated, cause mistrust, and in some instances create confusion of ideas or perception where none should exist. It must be understood that no guess work is allowable in science; and in the case of producing the correct

lap, a diagram of the angle of the crank is required to ascertain the lap when the stroke of the valve is not given. It may, of course, be argued that a combination of figures properly dealt with *must*, from natural laws, produce correct results, irrespective of the complication of the formula used. It cannot be denied, however, that simple diagrams greatly assist the student. Before proceeding further, it will be better to define the relative proportion of the circle commonly known as the travel of the valve.




Let it be presumed that a given slide has a travel of 6in., width of each opening or port to equal 1in., the outside lap will be half of the product or quotient, the ports being the divisor, and the travel the dividend. Then, from the

extremity of the circle denoting the travel, draw a line so that the areas of the segments shall be equal; from the points of contact set off the widths of the openings; the distance between, or length on the line, will be the total lap of the valve, and a line drawn at right angles with the line of construction or plane line will further tend to pourtray the proportion of lap and port (see the diagram). It will thus be readily understood that from the first point of contact to that of the second, must be the width of the port or opening, and from thence to the centre line must be the lap. Thus far, having briefly described the principle of the circle due to the eccentric, allusion must be made to the ratio of the length of the connecting rod to the stroke of the engine. This ratio affects the lap of the slide valve, when the point of cutting off is determined. This is perceptible on referring to a table of sines, or calculating the same from or by the length of the chord and the radius of the arc, or length of the rod. It may be more clearly understood by the large diagram introduced on pages 36 and 37.

This diagram is presumed to represent the centre lines of the angles of the crank and connecting rod during different portions of the stroke of the piston. The ratio of the stroke to the rod

is as 1 is to 2·5, hence the length of the chords of the arcs, which last are seen in dotted lines. It will be apparent that the versed sine previously noticed imperatively determines the angle of the crank in strict proportion to the speed or action of the piston. In the diagram now alluded to, the stroke of the piston is divided into six equal parts, as is the plane or centre line of the crank's path to correspond. Now let it be assumed that the piston has moved the crank $\frac{1}{6}$th of the stroke, which, for the purpose of calculation (the stroke) shall equal 24in., then $\frac{24}{6}=4$in. It will be seen that the crank is at 1, presume the piston to have forced the crank to 5, the difference in the angles of the crank and length of the arc passed through will be perceptible. It must be borne in mind, however, that the speed of the piston must have been unequal, *i.e.*, presuming the speed of the crank to be uniform on the arc of its travel or path. This, however, does not materially affect the mechanical arrangements of the engine, nor very little (if at all) lessens the power impressed on the piston. The length of the travel of the crank to that of the piston is, of course, due to the ratio between the circumference and the diameter of a circle. It may not be out of place to add that there is no rela-

tive value in the points of stoppage, so to speak, on the crank path to that of the piston; neither will the distances between the points on the former be equal, the variation being due, of course, to the length of the rod and crank. The proportions of the lap of the valve next become worthy of comment. As before stated, many valuable rules have been given to determine the correct lap of a valve for a given purpose. Now, in most, if not in all these rules, the travel of the valve is mentioned; which, in fact, is the most important feature of the formula. This is, of course, correct when the widths of the ports are not known, and the travel of the valve is assumed. Now the ports are generally determined before the valve and gear is proportioned; the rules, therefore, should be *seriatum*.

For the better purpose of descriptive illustration, let it be presumed that a pair of engines give unsatisfactory results, and it is agreed to alter the grade or cut-off of the valve—the steam or valve casing allowing an increase of stroke to a given extent; then the travel of the valve should undoubtedly be first noticed, and new valve gear introduced. To exemplify again: assume a pair of engines to be designed, the steam to be cut off at a given part of the stroke, and a given width

of ports imperative (which should be always first determined); the total travel of the valve could only be known, from or by the rule that the outside lap, plus width of opening = half travel of the valve. As before stated, the angle of the crank is of the most vital importance, and in no case whatever can a truthful travel of the valve be attained without taking it into consideration. To refer to the second diagram in this chapter, it will be remembered that allusion was made therein to the versed sine, which will be seen to equal the width of the steam port or opening. This fact is the only sound basis for a correct mathematical problem or diagram that can give the required result. Now, let it be required to find the lap of valve to cut off at one-sixth of the stroke of the piston; first, find the angle of the crank; next agree as to the width of the steam opening. The proper mode of proceeding further would be undoubtedly by calculation, but the diagram being so far constructed, and its relation to the value of the lap being imperative, a matter of simple geometry will prevent much confusion and doubt. Therefore find, mathematically, the radius of an arc, intersecting with the angle and plane lines of the crank (within its path); this intersection will determine a given length of chord. The versed

sine being given (with or without lead, as may be determined on), find if it agrees with that in the constructing diagram, presuming that the versed sines agree; the outside lap of the valve=radius of arc—versed sine. On referring to the first diagram this will be plainly understood; as to the necessity of this statement it may be added that if the angle of the crank be more obtuse, the radius alluded to will be reduced, assuming the same versed sine to be maintained. An argument may be raised that, in the absence of diagrams, a formula should be given to attain a result equal to the diagram. Now, in practice, engineers prefer to set out all the motions of the details of the engine; consequently, in the present case, a diagram answers every purpose better than a for-

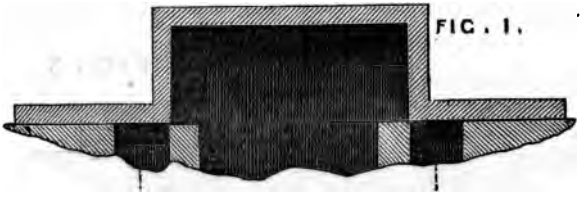
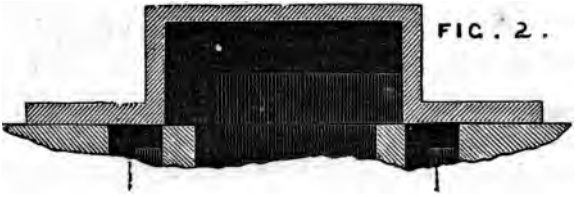


FIG. 1.

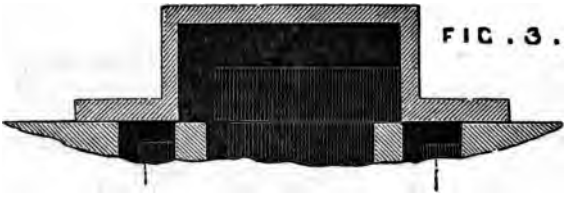
mula. Having thus far briefly explained the principles of obtaining the outside lap, attention will be given to Figs. 1 to 5.

The valve in Fig. 1 is intended to cut off at one-sixth of the stroke of the piston, or when the

angle of the crank is at the point 1 (on the first diagram). The dotted lines in Fig. 1 (as in the remainder) refer to the width of the opening caused by the valve, hence, in all cases, the half

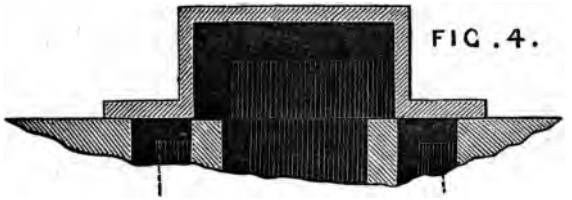



travel can be understood. The degree of expansion here given is perhaps rather higher than the general practice, hence the area of the cylinder is increased, the steam opening being reduced in width, but not in area, the length making good the requisition. Fig. 2 represents a valve and ports

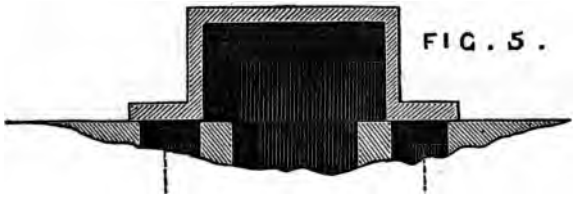



to cut off at one-third; on referring to the diagram the point 2 defines the angle of the crank. The lap in this case, also the opening, are both varied in relation to those for Fig. 1. The next,

Fig. 3, illustrates a valve and ports arranged to cut off at half-stroke, which is a good proportion, giving practical results. Refer to the diagram again, where the point 3 denotes the angle of




the crank. Fig. 4 is a valve and ports to cut off at two-thirds of the stroke. The angle of the crank can be seen, on noting the position of the point 4 in the diagram. Fig. 5 is introduced merely to illustrate the greatest extent steam should be cut off at. The openings in these three last examples are alike, hence the propor-




tion of lap can be readily understood. In this last figure the valve cuts off at five-sixths of the stroke of the piston; the angle of the crank can be seen on referring to the diagram. It must be

understood that the lead of the slide also regulates the travel of the valve. The angles for the eccentrics are determinable from or by the lead of the valve on the piston. The advance of the slide from the full stroke will then be equal to the width of the opening, minus lead. As this, however, may not be clear, the writer proposes to further elucidate the formula alluded to. Let it be assumed that the opening caused by a valve equals one inch, and the lead one-eighth of an inch, then seven-eighths of an inch *must be the advance of the valve from the full stroke when the piston is at the extremity of its path.*

Now, in order to make this still better understood, the following compilation, from the author's "Pocket-book of Practical Rules for Land and Marine Engines and Boilers," will assist:—

"The angle of the eccentrics in relation to that "of the crank will be thus produced:—

"First.—Describe on the centre of the crank "shaft a circle equal to the travel of the slide "valve.

"Secondly.—Draw the centre line of the crank "at full or half stroke.

"Thirdly.—The intersection of the circle (de-"noting the travel of the valve) with that of the

" centre line of the crank is the starting point to " ascertain the angles of the eccentric.

" Fourthly.—From the point of intersection " furthest from the crank set up a distance or " dimension equal to the width of supply steam " port or opening caused by the slide, minus lead " of slide.

" Fifthly.—At the point where this distance in- " tersects with the centre line of the crank, draw a " line at right angles to it, the intersections of " which, on the circle, will be centres of the ec- " centrics.

" Sixthly.—Lines drawn from the centre of the " cranks to the centres of the eccentrics will be " the angles of the latter."

It may as well be added that all the rules for the proportions of the valves and ports alluded to are compiled from the same work.

The difference required in the outside laps of a valve is more practically determined by diagrams than by calculation; due, of course, to the angles and versed sines. One potent fact should not be neglected, viz., *that the path of the crank is virtually that of the eccentric.* Thus, the arc passed through by the crank, at a given radius, must be that of the eccentric at the same radius, irrespective of the angles or relative positions of either.

## CHAP. V.

### THE EFFECT OF EXPANSION BY THE ACTION OF THE SLIDE VALVE.

THE economization of fuel is a matter of the highest importance to all, but more especially to the members of the commercial and scientific communities. The former gain in a pecuniary sense, and the latter in the further knowledge of natural laws. Now, to cause a reduction in the consumption of fuel, the evaporative powers of the boiler, and the mechanical arrangement of the engine must be well considered.

The principle of the action of engines of the present day may be said to be the same throughout. The proper distribution of the steam is a mechanical operation, and the results, bad or good, due to the failure or perfection of the means employed. The machinery for ships has become gigantic, and, therefore, requires due attention. In our navy, and also our mercantile fleet, many of the pistons of the engines weigh over five and a half tons. An imaginary conclusion, therefore, of the power required can be

readily formed. Steam, being very elastic, naturally admits of great compression—hence the gain in working or using it expansively. It is universally known that, the greater the pressure, the power will be increased in proportion. Now, let it be presumed that steam is required for a cylinder at the rate of 100 cubic feet per stroke of the piston, at a given pressure throughout, thus imparting to the piston a certain power. It is obvious that the given quantity of steam must be evaporated in rotation from the boiler; leakages, friction, reduced temperature, &c., being also considered.

Let it be assumed, also, that a given quantity of fuel is consumed in proportion to the power attained. Suppose it be deemed necessary to reduce the consumption of the fuel in the next example, or, rather, pair of engines. The first consideration of the improver—or, rather, the designer—would be, that the power must not be reduced. This, then, produces the idea that to increase the pressure of the steam, imperatively increases its elastic force. Thus, it is clear, that by increasing the pressure, a reduction in cubic contents of steam from the boiler is attained, requisite for each stroke of the piston; and, conquently, the evaporation required is lessened in

proportion. Now, it must be strictly understood that, to produce correct expansion, cubical capacity in the cylinder must be observed. Thus, for example, a cylinder of a given capacity for a certain pressure of steam throughout the stroke of the piston, must be enlarged in proportion to the requisite increase of pressure of the steam at the commencement of the stroke. To use steam economically is, to reduce the temperature of it as low as possible before it enters the condenser, and thus affect the time of condensation, producing thereby a better vacuum than if the temperature was higher. It should be remembered, also, that each volume of steam from the cylinder must be perfectly condensed before the succeeding volume enters. Having thus far briefly alluded to the principles of economy, let it be next considered what is the assumed action of the steam while in cylinder, *i.e.*, from the *entrée* to the point of exhaustion. Presume the piston to be at full stroke, or at the top or bottom of the cylinder, the slide valve should have a lead, or be open for a given portion. Now it must be noticed that the steam *was* admitted in the cylinder *before* the piston had completed its stroke, consequently compression, and partial if not a total stoppage of the *entrée* must have ensued. Trivial as this

may appear, yet in high velocities and full leads the indicator diagram shows a round corner, commonly known as cushioning. The lead, however, must not be discarded, as it acts as a spring between the piston and the ends of the cylinder, a matter of great essentiality in short strokes, and, as before stated, high velocities. To return to the piston at full stroke, presume the crank to have passed the dead centre; the steam will then act with force due to its admittance, so that it is not erroneous to assume that the greatest effect of the steam on the piston is, when the valve is at full stroke, or the port widest open for the supply. The piston, be it remembered, is now supposed to be exerting its utmost efforts; due, of course, to the action of the steam. Presume the valve to return and close the port at a given point of the stroke of the piston. At this point expansion commences, the piston being propelled after that by the elasticity of the steam; hence the gain in economy before alluded to. The time the steam is admitted into the cylinder is, while the valve is moving forth and back, or opening and closing the steam opening. The time the steam commences its actual propulsive power on the piston to that of its cut-off = width of opening $\times 2$—lead. The valve, when last alluded to, was

presumed to be at the edge of the port, expansion therefore being in full operation. Before the steam can exhaust, the valve must open the same port, but contrary in its action. The time ensuing between the point of cutting off the supply and the opening the port for the exhaust is, of course, due to the outside and inside laps. Therefore the time allowed for expansion = outside lap + inside lap. This will be better understood by presuming the valve to have no laps, it will then be obvious that if additional length be added to the valve, that increase must be observed. The time allowed for exhaustion is due to the speed or travel of the valve from a point to a point, or in principle as before stated for supply and expansion. Now, presuming the valve has covered the port, and travelled so that the inner part is at the inside edge of the port, it can readily be understood that exhaustion must ensue instanter with the action of the opening. The valve now moves forward and backward for a given length, occupying thereby a certain time. If there was no lap inside, it is obvious that when the valve was at half stroke the time for exhaustion would be while the valve was opening and closing the port or opening, on the opposite end of the cylinder, outside lap being also con-

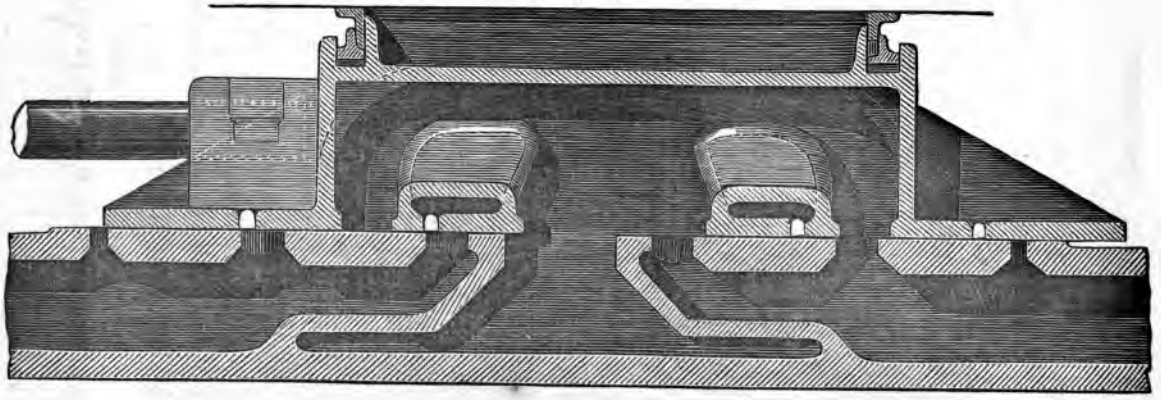
sidered. When there is an inside lap, the valve from the half stroke must move that sum or distance before the exhaustion commences; the valve then moves back due to the distance as before stated, and forward the same less the inside lap. The formula for time of exhaustion, therefore, will be thus, outside lap+width of opening caused by valve $\times$ 2.—[inside lap $\times$ 2.]

By these formulæ (time being considered) an almost perfect knowledge of the action of the steam can be attained, its elastic force and the volume consumed at each stroke of the piston known. There may be, perhaps, existing doubts as to the application of the rules now noticed, also their practical use; a little further elucidation will, therefore, not be out of place. For example, presume an engine of 50-horse power nominal; length of stroke, 2ft.; diameter of cylinder, 47$\frac{7}{8}$in.; utmost grade of expansion, $\frac{1}{6}$ of the stroke. Let it be assumed the travel of the valve to be 9$\frac{7}{8}$in. unalterable. The cubic contents of the cylinder will be deduced in the usual form $1800 \times 24 = 43200$ cubic contents in inches. The space occupied by the steam at $\frac{1}{6}$th $= \frac{43200}{6} =$ 7200, or $1800 \times 4 = 7200$. This, then, shows that the expansive powers of the steam must

emanate from a volume of steam whose capacity equals $\frac{1}{6}$th of the total contents of the cylinder. Now the amount of expansion is, of course, due to the capacity occupied after the termination of the *entreé* of the steam; to define, therefore, the amount of expansion time and space must be duly considered. The piston in the present example is presumed to have moved 4in., or $\frac{1}{6}$th of the stroke. The valve has closed the port, and has to travel the outside lap+inside lap, until the steam is released from the cylinder. To further demonstrate the present theory, time must be noticed. Let it be assumed that the engine, or rather the piston, moves at the rate of 160 strokes per minute — two strokes, of course, making a revolution. The valve, be it remembered, moves at the same speed, action only being now alluded to. Now, when the piston is at the extremity of its stroke, the valve has performed the major portion of its movement to the utmost point, and, consequently, in a given time moves in a contrary direction. The relative positions of the valve to the piston are due to the arcs passed through rotatively, each being the same in proportion. Circumferences of circles bear an equal relation to each other as their diameters; due, of course, to the constant number used.

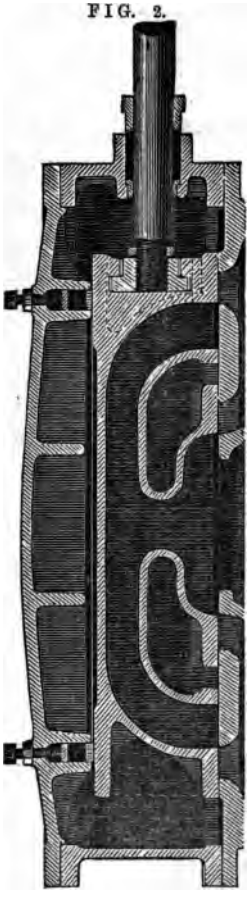
Thus, a circle 47⅞in. diameter equals 150·4in. in circumference; a circle 9⅞in. in diameter equals 31.02 in circumference. The proportion of the diameter will be as 1 is to 4·84; then $\frac{150{\cdot}4}{31{\cdot}02} =$ 4·84, proving thereby that the arcs passed through are the same in proportion lineally. To produce the amount of expansion by calculation, in the absence of diagrams, is a difficult matter perhaps, without accomplishing a correct result, due to the length of the connecting and eccentric rods, the versed sines of which bear strict reference to the action of the valve. It is, therefore, advisable to practically set out all the points of contact and requisition, in order to produce a perfect result. For example, when the piston is at full stroke, the eccentric is at a given angle, due to the width of opening or port and the lead determined or decided on. Now the crank moves through a certain arc, and the *eccentric virtually through the same*, or proportionately the same; but the distance on the plane line is entirely different; due, of course, to the angle of the chords. In the case of the crank the chord is at an angle, but that for the valve may be termed almost perpendicular. It is not argued that calculation *could* not produce a correct result, but

FIG. 1.



rather that diagrams *always* assist elucidation. Expansion produces economy, and the lap and lead of the valve, in proportion to the stroke of the engine, govern expansion.

FIG. 2.



It has been stated that by the introduction of two ports in the place of one, a reduction of one half of the travel is attained. Now, in order to still further reduce the stroke of the valve, three ports are sometimes adopted. It may be argued that by this arrangement the steam will be wire-drawn. In practice, however, this presumed evil is not apparent, and an almost perfect indicator diagram has been taken from cylinders fitted with three-ported valves. The illustration, Fig. 1, represents a valve arranged by Mr. C. Sells, chief draughtsman to the firm of Messrs. Maudslay, Sons, and Field. The valve is depicted at full stroke, hence the relative proportions of

the means for the supply and exhaustion can be readily understood. Valves, thus designed, have been fitted to many vessels with good results, in fact almost equal to the most perfect expansion valve and gear.

The illustration, Fig. 2, represents a slide valve and casing, as designed and constructed by the firm of Messrs. J. and G. Rennie. This example is perhaps the most graceful sectional form that can be conceived. The illustration being complete, further description will be superfluous.

## CHAP. VI.

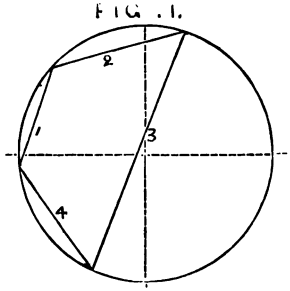
### THE DELINEATION OF THE PATH OF THE CRANK PIN.

The motion imparted to the slide valve is generally derived from two principles of action—vibratory and rotary. Now, when the former is the prime mover, the speed of the valve is the same throughout the stroke, or rather, if the motion is imparted by the piston, the motion of it and the valve would be equal. Rotary motion is more often adopted than any other for the transmission of power and action, and to the present day the small cranks and circular eccentrics are the prevailing means employed to impart the motion required for the slide valve. The speed of the crank and the eccentric are proportionately the same in theory and practice. The length of the connecting rod in all examples of rotary motion regulates the inequality of the speed of the sliding body; it must be remembered, however, that a reciprocity of motion can be attained by correct distribution of detail.

Before proceeding further with the definition of the better arrangement it will not be out of place to define the relation of the path of the

crank to the sliding motion imparted to the valve, also the controlment of the steam. It is doubtless universally known that, virtually, the crank path is divided into four distinct parts, also that for the eccentric. The proportions of these divisions are practically regulated by the grade of expansion agreed on to be maintained. Fig. 1 represents a crank path, the chords indicating the

FIG. 1.




divisional points, the relative proportion is therefore, readily understood. No. 1 is the chord of supply, No. 2 that for expansion, No. 3 relates to exhaustion, and No. 4 represents neutrality, or that portion of the stroke of the piston when the port on the exhaust side is covered, often termed compression, consequently, the piston for a period is devoid of pressure or vacuum. The length of the chord 1 is due to two causes, the grade of expansion and the length of the connecting rod. It will be noticed that the chord at the plane line

intersects with the circle slightly below the same, this last intersection is the angle that the crank assumes when the slide valve commences to open the port, and the vertical distance from the intersection to the plane line is due to the lead required. The upper point of intersection, as before explained, is subject to the curve assumed by the connecting rod from a point, or distance, on the plane line to the circle of the crank path. The length of chord No. 2 is regulated by the inside and outside laps of the slide valve. The expansion of the steam is now in full operation, and is released by the opening of the port on the exhaust side, hence the intersection of the chord at No. 3. The length of the last mentioned chord is more than any other, due to the traverse of the valve, or the time occupied in opening and closing the port for exhaustion. Now in the case of an increase of supply steam, the time for expansion and exhaustion would be lessened in proportion, it being remembered that the circle described by the crank pin cannot be increased or decreased for a given length of stroke of piston. The circle, as before stated, is divided into four divisions, and the alteration in the grade of expansion or length of connecting rod affects the whole proportionately. The concluding chord, No. 4, repre-

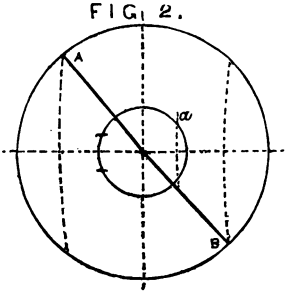
sents neutrality, or as before stated, that portion of the stroke of the piston where the vacuum and steam is cut off for a given period, commonly known as compression. It will be remembered that the chord of expansion is due to the outside and inside laps, *i.e.*, when the valve is at the edge of the supply side of the port, the valve has to travel forth until the inside edge permits exhaustion or destroys expansion. The valve is now at half stroke, plus inside lap; exhaustion, therefore, must ensue until the valve is in the same position, but travelling in a reverse direction. The position of the valve when terminating exhaustion will be half stroke, minus inside lap. It can thus be clearly understood that the length of chords for expansion, and that for compression, are equal in the example given; it may also be added that any variation in these two chords will depend on various causes, such as unequal laps and leads, &c., &c., but with certain arrangements both are equal. The fact of the compression being the same as the expansion, is of no vital importance; it is certain there would be a gain in maintaining expansion longer, and exhausting, till the supply commenced, and thus dispensing with compression; but the present motion of the slide valve would have to be dis-

carded, as an extreme unequal action would have to be attained.

The proportions of the path of the crank pin can thus be clearly understood. As at present arranged—compression, supply, and exhaustion, form one portion of the circle, and the remainder is occupied by exhaustion, the latter predominating. The diagram given relates to one revolution of the crank only, it being well known that each in principle is alike. There is, however a slight variation in practice, due to the versed sines of the rods, and inequality of the length of arcs passed through, in proportion to the propelling or sliding motion attained. The arrangement of the slide valves, in relation to that of the connecting rod, should be considered as to the attainment of equal action. The versed sines of the chords of the arc passed through by the crank pin, at each angle of the stroke, for a given supply of steam are unequal, also that for the eccentric, when the slide is at the same side of the crank as the connecting rod. This can be counteracted to a certain extent by short eccentric rods, and levers, reverse in action, and located to compensate for the inequality of the speed of the circular and sliding motions, the attainment of which being of great importance.

Fig. 2, represents the circle described by the crank pin for a given stroke of piston. The plane line is presumed to be the centre of the the engine, and the larger dotted arcs represent

FIG. 2.



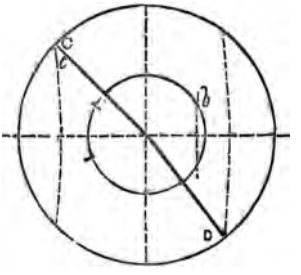

an even grade of expansion, on each side of the piston or at each stroke. Now it will be seen that on the crank (represented by the thick lines,) moving from the plane to the intersection at A, a given length of stroke of piston is produced, due, of course to the radius of the larger dotted arc. On the crank reaching to the intersection at B, the grade of expansion is reversed in action, but the same distance from the end of the stroke retained. Now, the difference in the lengths of the arcs passed through are due to the length of the dotted arcs, the radii of which are the connecting rod. The smaller circle *a* denotes the travel of the valve or the path of the centre of formation of the eccentric. The dotted

arc indicates the distance the valve must be from the edge of the port, when the piston is at full stroke, hence the angle or advance of the eccentric to that of the crank when on the horizontal line. When the crank is at A, the eccentric has passed through an arc equal in length proportionately. The two dots on the circle opposite the dotted arc indicate the angles of the eccentric when the crank is at the plane line and at B. Now it will be readily seen that the space between the dots is less than that of the intersections opposite; also the versed sine reduced. It is obvious that an unequal grade of expansion must ensue on the side of the circle at B, *i.e.*, if the laps of the valve are equal. It must also be remembered that, to increase the lap and retain the previous stroke of the valve is to destroy the lead; hence, the slight variation in the time, for supply steam at each stroke of the piston may be said to be endured rather than introduce a worse evil.

It is not here intimated that perfection of mechanism should not be attained, but rather to delineate and describe that most universal. The diagram now under question, illustrates the principle of the action of the crank and eccentric with the valve located on the same side of the crank shaft as the connecting rod, direct action in each case being maintained. It will be seen that the

arcs are all struck by the radii on the same side of the perpendicular line, hence the variation above alluded to. Now in order to counteract, or rather obviate, the imperfections now under notice, the versed sines of the chords indicating the partial stroke of the valve, must be equal on

FIG. 3.




each side of the perpendicular line. Fig. 3, represents a crank pin path, and that of the eccentric, to cut off at the same grade of expansion as Fig. 2. In the present case the arcs passed through are reversed to that of the former, the piston presumed to be moving in the same direction, but the locality of the connecting rod opposite to that of the slide valve. It will be noticed that the diameters of the eccentric paths are unequal; this inequality is due to the variation in the arc of the crank pin's passage during the forward and backward grades of expansion. When the crank has moved from the plane line to C, the eccentric has passed through an equal arc,

and the same relative motion occurs from the horizontal line to D. Now the length of the intersection at *b* can be seen, in proportion to that between the dots in the circle opposite. Suffice it to say, the nearer these two intersections agree in length or space between the same, the less variation will ensue in the grades of expansion at each double stroke of the piston. It will be remembered that the position of the pistons in Figs. 1, 2, and 3, are presumed to be alike; also the direction of the movement. It may as well be added that on reversing the action of the pistons, the travel of the valves would be effected. In some instances the greater travel of the valve is preferred, due to the causes alluded to. To attain a correct action for the slide valve, would be, to introduce cam motion, but the complication, and wear and tear, subject to high velocities often tend to discard the idea. Mitre gearing has been adopted with a cam on the motion shaft. This arrangement deserves more attention than the cam on the crank shaft, although complication may be said thereby to predominate. With steel cams and rods and perfect lubrication, there is no reason to doubt the durability of the same. The vital difficulty to overcome, is a perfect means of starting, stopping, and reversing.

## CHAP. VII.

### GENERAL OBSERVATIONS.

The preceding chapters have dealt with the dimensions to be attained, the cause and effect of the same, which of course are the primary considerations. The arrangement of the ports beyond the valve facing has not been alluded to, therefore, a few hints will not be out of place. Having determined the dimensions to be adhered to, the line of the valve facing should be first drawn,—a line at right angles will denote the centre of the exhaust port. Should a rib be required, half the width of the exhaust port will commence on each side of the same, a good proportion for which is, thickness of rib equals thickness of cylinder, a fitting strip being provided on each side. Presuming the width of the exhaust port to be drawn, the inside or narrow bar must next be depicted. Thus far having proceeded, the width of the supply port should be shewn. This completes the setting out of the ports for the ordinary single ported slide valve. Should the equilibrium type be required, the same process of construction as before, must be resorted

to. Having thus far proceeded, the large bar is added, after which the width of the port supply. The raised face beyond the port, need not be much wider than the narrow bar, for the just cause, that if the latter be sufficiently wide to render the valve connection steam tight, an increase of surface will be superfluous. Another reason is, that the least amount of surface in contact with the valve, the more perfect the connection, and the less the friction. Having thus far settled the proportions and representations of the same, set off the lap of the valve beyond the port, from this last point, the half travel plus the clearance, which will represent the commencement of the valve or steam casing. The width of the flange will depend on the diameter of the stud or bolt used, for which a good proportion is, half an inch to seven-eighths of an inch in diameter, and the pitch of the bolts eight times the diameter. The termination of the ports in the cylinder, is due to the length of the stroke and the depth of the piston, which will decide the distance between the ports, where entering the cylinder at the top and bottom. In some cases the piston is allowed to pass beyond or slightly cover the ports at each end of the stroke, so that the wearing surface may be equal through-

out. One important fact should be attended to, *i.e.*, the ports at the valve facing should have fitting strips on all sides. This matter is too often disregarded; it is generally overlooked that the fitting or shaping of the ports can be accomplished with greater facility, when provisions are introduced than when the same are absent. For oscillating engines, the setting out of the ports is much as that last described, but an additional supply port is introduced at right angles with those ordinarily arranged. This last mentioned port communicates with the steam passage, leading from the supply trunnion, that for the exhaust being oppositely located. In designing, or rather arranging the flange for the valve casing, it should be remembered that the same level must be retained on all sides, both for the better means of planing and making a perfect joint. In setting out the steam passages, care must be taken to increase rather than decrease the dimensions, those for the ports being now alluded to. Should the area of the passage (by accident or carelessness of design) be less than that of the port on the face, the latter is virtually reduced to the same size as the passage.

The proper means of ascertaining the depth of the valve will be to consider the width of the sup-

ply port allowed for exhaustion. In some instances an increase is deemed necessary to allow a more free exhaust, but in practice the width of the port is the vital part, as through it the exhaust steam must pass before it reaches the final or centre exhaust port. The areas of the steam side openings, in equilibrium valves, must each equal half the area of one steam opening caused by the valve. By this proportion it will be understood that sufficient steam can enter the openings in the valve, on each side of the same. In some examples these openings are contracted towards the centre, which may be said to be correct, as it allows a larger exhaust above the openings alluded to. The depth of the valve above the metal of the openings should equal the width of the supply port in the cylinder. The longitudinal distance between the steam openings in the valve, is due to the travel of the same, which must be duly considered to attain a correct action.

The steam passages of large cylinders are strengthened by ribs in a line with the cylinder, commencing at the valve facing and terminating with the passage itself. These ribs must be equally disposed; should two be used, the length of the port should be divided into three equal portions; if one rib be required, two divisions

will only be requisite. It is not imperative that the edges of the ribs should be on a level with the valve facing, but rather on a level with the lower edge of the fitting strip, which will be better both for planing and fitting. When the valve facing is at an increased distance from the cylinder, the ports should be arranged as before mentioned, but the passages as direct as possible, or at an angle rather than in a line with the body of the cylinder. The position of the valve on the cylinder for horizontal engines will, of course, greatly affect the friction. Should the locality be the top of the cylinder, the fricton will be greatest; if at the bottom or the underside, the least; and if at the side of the cylinder, a happy medium will be maintained. This last mentioned position is most universal, due, of course, to the cause alluded to; and also as the best means of access for repair and renewal of the surfaces. Large valves have a guide at the bottom side, to prevent a lateral strain on the rod.

It will doubtless have been noticed that, at the conclusion of the first chapter, allusion was made to the means generally adopted for reducing the friction on the valve facing, caused by the pressure of the steam acting on the back. It is now proposed to further treat of this important portion of

the present subject, so that the better means may be fully appreciated. With the springs and recessed ring, adjustment is only available by entire disarrangement, but with good workmanship and careful treatment it may, nevertheless, be deemed admissable. In order to admit of ready adjustment, without disarrangement, the ring encompasses the hollow projection at the back of the valve, instead of being recessed in the same; four to eight studs, with collars bearing under the ring, act as a means of adjustment. The said studs are prolonged through the provisions on the ring, and screwed into bosses at the sides of the valve. Access is separate to each stud by provisions arranged on the casing cover. A separate ring, contained in that first mentioned, acts against the inside of the cover, and thus prevents the action of the steam on the back of the valve. In some instances gasket is used between the two rings, but with super-heated steam, undoubtedly springs of a spiral or wave-like form are better. In the place of the ring encompassing the projection on the valve, or being recessed in the same, some authorities prefer the ring to be recessed in the cover, which also, of course, accomplishes the desideratum. *See page* 57, *Fig.* 2.

A central communication with the exhaust

steam pipe or the condenser, causes a vacuum within the ring surface, and thus reduces the friction on the valve facing. Stop cocks are provided and sometimes fitted in the cover of the casing, so that a plain external is maintained.

The modes of securing the rod in or to the valve are not much varied, in some cases the rod passes through the valve, and is secured at each end of the same by nuts, and thus attain adjustment; also another mode is, to recess a nut in the front part of the valve, and screw the rod in the same; this is the best connection yet introduced, a fact alluded to in the first chapter. Two rods to each valve are now becoming more universal, but the mode of connection is the same as the last mentioned. The stuffing box and gland for the rod should be properly considered; a good proportion for the diameter of the stuffing box is, diameter of rod $\times 1{\cdot}5$. Depth of stuffing box = diameter of the same $\times {\cdot}6$ to ${\cdot}5$, while in some examples the depth will be found to equal the diameter. The depth of the gland is not of much importance, but a good proportion is attained by depth of stuffing box $\times {\cdot}75$ to ${\cdot}6$. The diameter of the studs for adjusting the gland is not imperative, half an inch to three-quarters of an inch being often adopted, two studs only being used. The

thickness of the flange of the gland generally equals the diameter of the bolt. With reference to the position of the securing flange of the casing, some makers prefer an outside connection, while others adopt an inside flange. If external appearance be considered as worthy of notice, the latter is the better. The cover of the casing should be ribbed either inside or outside, the latter being the most simple, while the former presents the better exterior. *See Fig.* 2, *page* 57. Having thus far duly considered the best mode of designing the slide valve, steam ports, and casing, allusion will now be made to the strains imposed by the steam. The casing, be it remembered, may be termed a box, within which a certain force is exerted. Now, on the steam being admitted, each portion is operated on by the same. The flanges are secured by the studs, hence the latter are subject to a tensile strain. The sides and ends of the casing are subject to a tensile and lateral or bulging strain, the former predominating. The cover and front portion of the casing are exposed to the greater effect of the pressure within the same, due, of course, to the surface exposed, hence the cause for ribs, &c. The action of the steam on the valve is greatly counteracted by the curved form

it generally assumes, hence the thickness of the metal may be considerably less than that for the casing. A good proportion is three-eighths to three-quarters of an inch for the thickness of the body part of the valve, and five-eighths to seven-eighths of an inch for the body part of the casing. The flanges in each case should be slightly thicker than the body part. The thickness of the ribs should be less than that of the body as the former are not subject to the full pressure of the steam.

In designing the slide valve, whether the ordinary or equilibrium kind, strict attention should be given to the speed of the piston, pressure of the steam, grade of expansion, length of the connecting rod, lead of valve on the piston, &c., matters which have been explained in detail. The only excuse for again reverting to these important facts is, to impress on the non-conversant, that, of all the details of the steam engine, the most important is the "SLIDE VALVE."

FINIS.

# INDEX.


| | PAGE |
|---|---|
| Action of the slide valve | 48 |
| Action of the steam | 51 |
| Action of the valve to cause exhaust | 52 |
| Action of expansion | 62 |
| Action of the steam in the cylinder | 35 |
| Advantage of double ported valves | 25 |
| Antecedents of the slide valve and ports in the cylinder | 5 |
| Angle of eccentric | 46 |
| Area of cylinder | 8 |
| Area of ports | 9–10 |
| Arrangements of the ports | 10 |
| Area of steam ports | 10, 13 |
| Area of exhaust port. | 14 |
| Arrangement of equilibrium valves | 25 |
| Arcs dotted | 37 |
| „ „ | 64 |
| „ „ | 66 |
| Arcs of grade of expansion | 64 |
| „ „ „ | 66 |
| Area of passage | 70 |
| Area of steam openings | 71 |
| Bar, width of | 14 |
| „ „ | 23 |
| „ consideration of. | 30-1 |
| Bolts, diameter of | 69 |
| „ pitch of | 69 |
| Cam motion | 67 |
| Casing steam | 69 |
| Casing bolts | 69 |

PAGE

Casing, cover of . . . . . 75
„ flange of . . . . . 76
„ thickness of . . . . . 76
Casing flange, position of . . . . . 75
Casing strains, inside . . . . . 75
Chord of supply . . . . . 60
Chord of expansion . . . . . 60
Chord of exhaustion . . . . . 60
Chord of neutrality . . . . . 60
Compression, chord of . . . . . 60
„ course of . . . . . 62
Condenser, communication with . . . . . 74
Connecting rod, locality of . . . . . 66
Common slide valve . . . . . 12–13
„ „ Rules. . . . . 14
Crank path . . . . . 55
Crank pin path, delineation of . . . . . 59
„ divisions of . . . . . 60
Crank, position of . . . . . 64
„ „ . . . . . 66
„ „ . . . . . 37
Cushioning, cause of . . . . . 18
Cylinder, capacity of . . . . . 50
„ area of . . . . . 8
„ „ . . . . . 13

Delineation of the eccentric travel . . . . . 39
Diagrams of common slide valves . . . . . 12–13
Diagram of exhaust relief slide valve . . . . . 19
Diagrams of equilibrium double ported valves . . 26-7
Diagram of ratios of connecting rod to crank . . 37
Diagram of ratio of port to lap . . . . . 38
Diagram indicator . . . . . 51
Diagram of laps of valves . . . . . 43, 45
Diagram of Sells' three-ported valve . . . . 56
Diagram of Rennie's valve . . . . . 57
Diagram, indicator . . . . . 57
Diagram of the crank pin's path . . . . . 60
Diagram of grades of expansion and locality of details. 64
„ „ „ „ . 66

PAGE
Division of crank path . . . . . 60

Economical use of steam . . . . . 36
Eccentric, travel of . . . . . 39
„ angle of . . . . . 46
„ „ . . . . . 55
„ angles of . . . . . 65, 67
Eccentric rods, location of. . . . . 63
Eccentric paths, cause of variation of . . . 66
Exhaust, consideration of. . . . . 71
Exhaustion, time of . . . . . 52–3
„ chord of . . . . . 60
Exhaust relief valve. . . . . 18–19
„ „ rules for . . . . . 18
Equilibrium and double ported valve . . . . 24
„ „ . . . . 25, 27
„ „ rules for . . 29
Expansion, time of . . . . . 52
„ difference of . . . . . 65
„ effect of . . . . . 48
„ grade of . . . . . 64
„ chord of . . . . . 60
„ operation of . . . . . 61

Friction of valves . . . . . 5
„ „ reduction of . . . . 6, 11, 73
„ „ rule to attain the amount of . . 6

General observations. . . . . 68

Indicator diagrams . . . . . 22
„ „ . . . . . 57
„ „ . . . . . 51
Illustrations.—Slide valves . . . . . 12
„ „ . . . . . 13
„ „ . . . . . 19
„ „ . . . . . 26
„ „ . . . . . 27
„ Angles of crank and connecting rod . 37
„ Eccentric path . . . . . 38

PAGE

ILLUSTRATIONS.—Lap of valve . . . . . 43
" " . . . . . 44
" " . . . . . 45
" Sell's three-ported valves . . . 56
" Rennie's valve. . . . . 57
" Division of crank path . . . 60
" Grades of expansion and locality of detail . . . . . 64
" " " 66
" Ratio of connecting rod to crank . 36-7

Lap, rule for attaining the . . . . . 7
" " " . . . . . 42-3
Lap of valve . . . . . 14
" " . . . . . 16–17
Lap, introduction of. . . . . 23
Laps, unequal . . . . . 34
Lap, correct amount of . . . . . 35
Lap regulates the cut off of the steam . . . 35
Lap, proportion of to port. . . . . 38
Lead, value of. . . . . 46
" " . . . . . 51
Lead of valve . . . . . 50
Lead of valve, effect of . . . . . 6-7
Lead of valve exhaust . . . . . 22
Lead of steam . . . . . 22
Levers, locality of . . . . . 63

Openings in valves, particulars of . . . . . 71
Outside lap, value of . . . . . 36-7
" proportion of. . . . . 38
" mode of attaining . . . . . 39-43
Overlap of valve . . . . . 33
Overlap, rule for . . . . . 33

Path of crank, division of . . . . . 60
Path of crank . . . . . 47
Path of eccentric . . . . . 47
Path of crank and eccentric . . . . . 55
Piston, travel of . . . . . 69

PAGE

Pistons, position of . . . . . 66
„ reversing in action . . . . . 67
Ports, number of . . . . . 57
„ arrangement of . . . . . 68
„ width and consideration of . . . . 68
„ setting out of . . . . . 7
„ „ . . . . . 68–9
„ fitting of . . . . . 70
„ setting out for oscillating engines . . . 70
„ purpose of . . . . . 8
Position of cranks . . . . . 37, 40
Proportion of valves and ports . . . . . 12
„ „ . . . . . 14
„ „ . . . . . 16-17
Proportion of outside and inside laps. . . . 17
Proportions of ports. . . . . 18
„ „ . . . . . 21
Proportion of crank path, divisional . . . . 60
„ „ . . . . . 63
Proportions of equilibrium valve . . . . 28, 32
Proper mode of attaining the correct amount of lap . 35
Proportion of lap to port . . . . . 38

Ratio of ports . . . . . 22
Ratio of ports . . . . . 32
Ratio of connecting rod to crank . . . . 36-7
Ribs in cylinders . . . . . 71-2
Rule for diameter of valve rod stuffing box . . 74
„ „ „ „ depth . 74
„ „ „ „ depth of gland 74
Rules for diameter of studs of gland . . . . 74
Rule for thickness of slide valve and casing . . 76
Rules for equilibrium and double ported valves . . 29
„ „ „ . . 32
Rule for action of steam . . . . . 51
Rule for steam supply . . . . . 51
Rules for time of expansion . . . . . 52
„ „ exhaustion . . . . . 53
Rule for over lap . . . . . 33
Rules for equilibrium valve . . . . . 34

PAGE

Regulation of the steam by the lap . . . . . 35
Rule for calculating the friction on the valve facing . 6
Rule for attaining the correct amount of lap . . 7
Rules, diversity of . . . . . 7
Rule for area of ports . . . . . 9–10
Reduction of pressure on the valve . . . . 11
Rule for width of bar . . . . . 14
Rule for width of exhaust space in valve . . . 14
Rule for width of exhaust space in cylinder . . 14
Rule for common slide . . . . . 14
„ exhaust relief slide valves . . . . 16
„ „ „ . . . . 18
„ „ „ . . . . 20–21

Shape of the ports . . . . . 10
Slide, position of . . . . . 63
Slide valve, motion of . . . . . 59
„ thickness of . . . . . 76
„ importance of . . . . . 76
Slide valves . . . . . 12
Steam port area . . . . . 13
Steam, elasticity of . . . . . 49
„ spare . . . . . 48–9
„ economical use of . . . . . 49
„ admission of . . . . . 49–50
„ stoppage of . . . . . 50
„ action of . . . . . 51
„ expansion of . . . . . 51
„ time of power . . . . . 51
Steam economy . . . . . 50
Steam pressure to be considered. . . . . 8
Steam pressure . . . . . 9
Studs for valve rod gland, diameter of . . . 74
Supply, chord of . . . . . 60

Time for steam supply . . . . . 51
Time for expansion . . . . . 52
Time for exhaustion . . . . . 52–3
Travel of valve. . . . . 42
„ „ . . . . . 67

PAGE

Vacuum, cause of . . . . . 50
Valve, Sell's . . . . . 56
„ Rennie's . . . . . 57
„ travel of . . . . . 67
„ correct action of . . . . . 67
„ clearance of . . . . . 69
„ depth of . . . . . 70-1
„ position of . . . . . 72
„ guide for . . . . . 72
„ projection at back . . . . . 73
Valve friction reduced . . . . . 6-11-72-3
Valve packing and spring rings . . . . . 73
Valve side openings, area of . . . . . 71
Valve rod . . . . . 74
„ mode of securing . . . . . 11
Valve rod stuffing box, diameter of . . . . . 74
„ „ depth of . . . . . 74
Valve rod gland, diameter of . . . . . 74
„ depth of . . . . . 74
„ diameter of bolts . . . . . 74
Valve rods, number of . . . . . 74
Valves, common . . . . . 12–13
„ relief exhaust . . . . . 19
„ „ . . . . . 21
„ equilibrium and double ported . . . 24
„ „ „ . . . 26–27
„ sizes of . . . . . 24
„ friction of . . . . . 24
„ double beat . . . . . 24
„ arrangement of . . . . . 25
„ travel of . . . . . 42
Width of bar . . . . . 14
Width of exhaust space in valve . . . . 14
Width of port . . . . . 16, 18, 21, 29, 32
Width of exhaust opening . . . . . 18, 21
Width of exhaust space in valve . . 18, 21, 29, 32
Width of supply opening . . . . . 21
Waste of steam, cause of . . . . . 35

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*Plate* 2. Details of High Pressure Engine, 12 H.P.; Scale 3-in. = 1 foot.

Cylinder, Piston, Slide Casing, Slide Valve and Rod, Connecting Rod, Piston Rod and Guide Block, Eccentric and Band, Slide Connecting Rod, Disengaging Socket, Eccentric Rod, Weigh Shaft, Lever Stud Bracket, Slide and Pump Levers, Main Lever, Starting Handle, Ball Governor, in detail, Governor Standard and Pully.

*Plate* 3. Details of High Pressure Engine, 12 H.P.; Scale 3-in. = 1 foot.

Main Framing, Fly-Wheel, Crank and Shaft, Feed Pump, Disengaging Socket, Link and Pin, Governor, Mitre Gear, Holding down Bolts, Studs, Plates, &c. &c.

*Plate* 4. Details of High Pressure Engine, 12 H.P.; Half-size.

Feed Pump, Valves and Box, Relief Valve, Safety Valve, Starting and Governor Valves, Engine Feed Cock, Boiler Front Doors, Fire Bars, and Supports.

*Plate* 5. Boiler and Settings, 12 H.P.; Scale ¾-in. = 1 foot; 4 Views.

*Plate* 6. Patent Antifriction Trunk Engine, 6 H.P. Invented by the Author, Burgh and Cowan's Patent. Arrangement, 4 Views; Scale 1½-in. = 1 foot; Details—Scale, 3-in. = 1 foot.

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*Plate* 8. Details of One Marine Engine, 200 H.P.; Condenser and Air Pump, Feed and Bilge Pumps, Cylinder, Equilibrium Slide Valve and Casing, Main Frame, Connecting Rod,—Scale 2-in. = 1 foot; Guide Block and Cross-head—Scale 2-in. = 1 foot; Discharge and Suction Valves for Air Pump, 8-in. = 1 foot.

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*Plate* 10. Details of One Marine Engine, 200 H.P.; Scale 3-in. = 1 foot.

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*Plate* 11. Scale 3-in. = 1 foot; Expansion Valve Gear, Throttle Valve, and end of Thrust Shaft for Marine Engine, 200 H.P.

*Plate* 12. Oscillating Engines for the Paddle Wheel, 150 H.P. Collectively; Scale 1-in. = 1 foot; 7 Views

*Plate* 13. Paddle Feathering Wheel, 75 H.P.—Scale 1-in. = 1 foot; 2 Views.

*Plate* 14. Details of Paddle Wheel, 75 H.P.

Paddle Centre, Plummer Block, Eccentric Shaft and Bracket Lever Arms, Rings, Stays, Rods, Bolts and Pins, Float Clamps, Arm Stays and Floats,—Scale 8-in. = 1 foot.

*Plate* 15. Griffith Patent Screw Propeller, with Lifting Frame. Scale 1½-in. = 1 foot; 5 Views.

*Plate* 16. Arrangement of a Pair of Marine Steam Engines of 900 H.P., for the Imperial Ottoman Iron-Clad Frigate, "Sultan Mahmoud," constructed by Ravenhill, Salkeld, & Co., London, 1864. Scale ½-in. = 1 foot.

*Plate* 17. Marine Boiler, with Superheating Tubes, Scale 1-in. = 1 foot; 6 Views.

*Plate* 18. Ordinary Condenser and Air Pump for Screw Marine Engine, 200 H.P., and Oscillating Paddle Engine, 75 H.P.; 8 Views.

*Plate* 19. Surface Condenser, 200 H.P. — Scale 1½-in. = 1 foot; 4 Views,

*Plate* 20. General Arrangement of Engines, Boilers, Shafting and Screw for a Despatch Boat, 100 H.P. Scale ½-in. = 1 foot; 5 Views.

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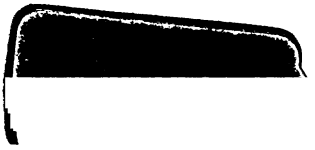
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